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COMPLETE SPECIFICATION

Improvements in or relating to Variable Speed Torque Transmission Arrangements

I, MICHAEL WIENAND, of Siegburg, An der Herrenwiese, Germany, a German citizen, do hereby declare the invention, for which I pray that a patent may be granted to me, and the method by which it is to be performed, to be particularly described in and by the following statement:—

The present invention relates to a variable speed torque transmission arrangement of the kind in which rotatable transmission means, preferably in the form of a discoid body, traverses a circular path, variation of the speed of rotation of the rotatable transmission means being obtained by effecting variation of the eccentricity between the axis of rotation of the rotatable transmission means and the axis of the circular path.

The invention has for its object to provide an improved construction of variable speed torque transmission arrangement of the above kind, the construction providing for positive rotation of the rotatable transmission means.

Accordingly the present invention provides a variable speed torque transmission arrangement of the kind hereinbefore set forth characterised by the provision of rotatable transmission means having a plurality of guide slots evenly spaced around a pitch circle and extending radially in relation to said pitch circle, and guide means located in each guide slot, the said guide slots and guide means co-operating to impart rotational movement to the rotatable transmission means about an axis coincident with the centre of the pitch circle in a direction opposite to the direction of the traversing movement of the rotatable transmission means around the circular path.

The transmission arrangement may for example comprise in combination drive shaft means, a first transmission member formed with a set of guide slots evenly spaced along a circle having a centre, the guide slots extending in substantially radial direction; eccentric means connecting the first transmission member with the first shaft means so that during rotation of the first shaft means the centre of the circle and the first transmission member move about the axis of the first shaft means; a set of evenly spaced guide members respectively engaging the guide slots and being movable in the same; a second transmission member supporting the guide members arranged in a circle having a centre; means supporting one of the transmission members for turning movement about the respective centre of the circle associated with the one transmission member whereby the one transmission member turns about the respective associated centre in a direction of rotation opposite to the direction of rotation of the first shaft means; second shaft means, one of the shaft means being a drive shaft means; and means connecting the one transmission member to the second shaft means.

According to a preferred arrangement of the present invention, the second transmission member has circular guide means extending around a circle having a centre, and a set of one-way coupling means is respectively connected to the guide member and are mounted in the circular guide means. The one-way coupling means block movement of the guide members in the direction of rotation of the drive shaft means.

Preferably, the eccentric means include at least one adjustable member supporting the first transmission member and being movable between a plurality of positions for varying the distance between the centre of the first transmission member and the axis of the drive shaft means. Operating means are connected to the adjustable member of the eccentric means for moving the same. Thereby the eccentricity is adjusted, and consequently the speed of the driven shaft is gradually varied.

As will be explained in greater detail herein-

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after, the ratio of the transmission according to the present invention depends on the ratio between the diameter of the circle along which the guide members are arranged, and on the eccentricity of the first transmission member with respect to the axis of the drive shaft means.

Since the first transmission member is eccentrically arranged, it is advantageous to provide counterbalancing means which are supported on the adjustable member of the eccentric means for balancing the weight of the first transmission member in each of the positions of the adjustable member. Whenever the first transmission member is shifted by operation of the adjustable member of the eccentric means, the eccentricity of the counterbalancing means is correspondingly adjusted so that the rotary members of the transmissions run quietly.

Accordingly to one embodiment of the present invention the second transmission member is a stationary member, and the driven shaft means is operatively connected to the first transmission member for turning movement with the same. For example, the driven shaft can be connected by universal joint means to the first transmission member. According to a preferred arrangement, however, the drive shaft and the driven shaft are coaxial. According to another embodiment of the present invention, the first transmission member is connected for rotation to a corresponding transmission member which is also formed with slots, these slots are engaged by guide members corresponding to the previously described guide members and associated with a rotary member which is connected for rotation to the driven shaft. By this arrangement the ratio of the transmission is further reduced.

According to a further embodiment of the present invention, the first transmission member is an annular member having a slotted inner circular rim engaged by the guide members of the second transmission member which is directly connected to the driven shaft. In this embodiment two eccentric means are associated with the annular transmission member, and are driven by parallel shafts which are connected by gear means to the drive shaft. The counterbalancing means are also mounted on eccentric means associated with the two parallel shafts, and are annular members corresponding to the annular first transmission member.

In all embodiments of the present invention, the adjustable members of the eccentric means are preferably zig-zag-shaped members which, when shifted in axial direction by suitable operating means, produce a shifting of the transmission member and of the counterbalancing means in radial direction.

In order that the invention may clearly be understood and carried into effect several embodiments of the same will now be described by aid of the accompanying drawings in which:—

Figure 1 is a schematic end view illustrating the principle on which the operation of an arrangement according to the present invention is based;

Figure 2 is a schematic sectional view taken on line II—II in Figure 1;

Figure 3 is an end view of one embodiment of the present invention;

Figure 4 is a sectional view of the arrangement illustrated in Figure 3;

Figure 5 is an axial sectional view of one embodiment of the present invention;

Figure 5a is a front view of a detail;

Figure 5b is a side view of the element illustrated in Figure 5a;

Figure 5c is a fragmentary sectional view taken on line Vc—Vc in Figure 5;

Figure 6 is a fragmentary cross sectional view taken on line VI—VI in Figure 5, some elements being shown in a side view for the sake of clarity;

Figure 7 is a schematic isometric sectional view of an arrangement illustrating the principle of another embodiment of the present invention;

Figure 8 is an axial sectional view of an embodiment of the present invention in accordance with the principle illustrated in Figure 7;

Figure 9 is a cross sectional view taken on line IX—IX of Figure 8;

Figure 9a is an isometric view of a detail shown on an enlarged scale;

Figure 10 is a fragmentary cross sectional view illustrating a detail of an arrangement shown in Figure 8;

Figure 10a is a fragmentary sectional view taken on line Xa—Xa in Figure 10;

Figure 11 is a side view of an eccentric means viewed in the arrangement of Figure 8 for varying the transmission ratio;

Figure 11a is a side view illustrating the device shown in Figure 11 in a different operational position;

Figure 11b is an end view of the device shewn in Figure 11a.

Referring to the drawings, Figures 1 and 2 illustrate the principle of a transmission of the kind with which the present invention is concerned. A circular disc 1 having a diameter d traverses a circular track formed by an annular member 2 having a diameter D. The disc 1 is turnably mounted on a crank shaft 3. The eccentricitv e of the crank portion with respect to the main axis of rotation of the crank shaft 3 is half the difference between the greater diameter D of member 2 and the smaller diameter d of the member 1, which may be expressed as follows: $D-d=2e$.

When the crank shaft is rotated, the centre of the disc 1 moves in a circle having a radius e about the axis of the crank shaft 3. At the

same time, the circular periphery of the disc 1 moves along the circular track 2 and may be considered to be in frictional engagement therewith. Consequently the disc 1 rotates in the direction of the arrow Q, that is in counter-clockwise direction when the crank shaft 3 rotates in clockwise direction. Each point P of the periphery of the disc 1 moves along a hypocycloid, as indicated by dotted lines. The disc 1 completes a revolution before having rolled along the entire circular track 2. Consequently the disc 1 rotates about its centre at a rotary speed which is different from the rotary speed of the drive crank shaft 3. Universal joint means 4 and 5 connected by a link 6 transfer the rotary movement of the disc 1 to a driven shaft 7, and it will be understood that thereby a transmission having a predetermined ratio is obtained. In order to counter balance the eccentrically arranged mass of the disc 1, a pair of counterbalancing weight means 8 and 9 are mounted on portions 10 and 11 of the crank shaft 3 at a phase lag of 180° with respect to each other and to the disc 1.

The ratio of the transmission, that is the ratio between the rotary speed of the disc 1 about its centre and the rotary speed of the driving crank shaft can be expressed by the equation: $\frac{D-d}{D}$. This equation corresponds to the equation: $\frac{2e}{D}$.

It will be apparent that the ratio of the transmission can be changed by varying the eccentricity e, and such arrangement is embodied in the construction illustrated in Figures 3 and 4.

For a proper operation of a transmission operating according to the principle illustrated in Figures 1 and 2, it is essential that the rolling disc rolls with its periphery without any slippage. If only a frictional engagement is provided, as schematically indicated in Figures 1 and 2, it is impossible to transfer a substantial torque.

Referring now to Figures 3 and 4, in accordance with the present invention, the circular track is constituted by a set of evenly spaced guide members 12 which are arranged in a circle having a diameter corresponding to the diameter D as explained above. A first transmission member 16 corresponding to the disc 1 with reference to Figure 1, is provided along the circular periphery thereof with a set of evenly spaced radial slots receiving the guide members 12 which are shown to be pins in Figures 3 and 4. The first transmission member 16 is driven by a drive shaft 17 having a prismatic end portion 18 projecting into a corresponding prismatic slot 19 of an eccentric means 19a. Member 19a has a circular periphery corresponding with an oppositely located circular inner surface of member 16. Between the two annular surfaces balls are arranged so that a relative turning between the eccentric member 19a and the transmission member 16 is possible. When the transmission member 16 is shifted, the eccentricity of the transmission member 16 with respect to the axis of the drive shaft 17 is changed. The slots 15 are arranged along a pitch circle whose centre corresponds to the centre of the disc 1 in the arrangement explained with reference to Figure 1. The ratio of this transmission is determined by the equation: $\frac{2e}{D}$. In this equation D is the diameter of the circle along which the pins 12 are located, and e is the distance between the axis of the shaft 17 and the centre of the circular rim of the eccentric member 19a. In accordance with the present invention, the pins 12 are mounted in oneway coupling means which are arranged in a circular groove of a second stationary transmission member 13. The one-way coupling means permit movement of the pins 12 only in the direction of rotation of the first transmission member 16, and block movement of the pins 12 in the direction of rotation of the drive shaft 17 which, as previously described, is opposite to the direction of rotation of the transmission member 16. Universal joint means 20 connect the transmission member 16 with the driven shaft 20a, in order to permit a connection between transmission member 16 and the driven shaft in all adjusted positions of the eccentric means 18, 19.

Due to the positive engagement between the pins 12 and the slots 15, the transmission according to the present invention is capable of transmitting high torques, while the transmission ratio can be gradually and infinitely adjusted by shifting the eccentric means. Due to the fact that the guide pins 12 are mounted for movement along a circular track only in one direction, a positive transmission of force is achieved, while clamping or binding of the pins 12 in the slots 15 during operation and during adjustment of the eccentricity is safely prevented.

Figs. 5, 5a, 5b and 5c illustrate an embodiment of the present invention in which the principle of operation described with reference to Figures 1—4 is embodied. The transmission is arranged in a housing including one end portion 21, a centre portion 22 and a cover portion 23 which are attached to each other by screw bolts 24 and 25. A drive shaft 26 has one end of prismatic shape 27 provided with grooves, best seen in Figure 5c, in which wedge-shaped members 28 are mounted for movement in axial direction. Drive shaft 26 is mounted in ball bearings 29, which are arranged in the housing portion 21, and in ball bearings 29a arranged in the centre portion 22 of the housing. The wedge shaped members 28, and the associated grooves in the shaft 26

extend along the entire prismatic portion 27 of shaft 26, and beyond that into the cylindrical portions of the shaft 26. Projections 32 are provided at the ends of the wedge-shaped members 28 and engage corresponding recesses in a sleeve 30 which is mounted in ball bearings 31. Consequently a shifting of sleeve 30 in axial direction effects a shifting of the wege-shaped members 28 along the shaft 26. The wedge-shaped members 28 have outer surfaces including three wedge-shaped faces 28*a*, 28*b*, and 28*c*. The wedge face 28*a* is located in a corresponding prismatic slot of an eccentric disc 33 which is mounted by means of a ball bearing 33*a* in the transmission member 34, so that when shaft 26 is turned, the eccentric disc 33 turns therewith and produces a movement of the transmission member 34, with the centre of the transmission member 34 moving in a circle about the axis of the shaft 26. The transmission member 34 corresponds to the disc 1 described with reference to Figure 1 and to the disc 16 described with reference to Figure 3. The transmission member 34 is provided along the circular periphery thereof with a set of evenly spaced slots 34*a* which are arranged in a circle. Guide pin members 35 have prismatic centre portions slidably guided in the slots 34*a* of the transmission member 34. The guide pin members 35 are arranged in a circle, and are movable in the direction of rotation of the transmission member 34, but blocked in the direction of rotation opposite thereto, which is the direction of rotation of the drive shaft 26. To obtain such blocking of the guide members 35, each guide member is connected by a one-way coupling means to a stationary transmission member. In the embodiment of Figure 5, the one-way coupling means are slip coupling means, best seen in Figure 6. The cylindrical ends of the guide pin members 35 are mounted in a pair of coupling members 37, which are arranged in circular grooves 36 of a stationary transmission member which includes two parts 36*a* and 36*b*. Each slip coupling member 37 has a portion 38 provided with wedge-shaped walls co-operating with balls 39 which are urged by spring means 40 into the narrower portion of the wedge-shaped spaces defined between the portion 38 of the coupling member 37 and the walls of the respective groove 36. The one-way slip coupling means 37, 38, 39, operate in a well-known manner to permit movement of the coupling means 37 and of the guide pin members 35 only in one direction, whereas movement in the opposite direction is blocked by the coupling means 37, 38, 39. Movement of the guide members 35 is blocked in the direction of rotation of the drive shaft means 26, and consequently the first transmission member 34 is forced to roll on the guide members 35, and more particularly on the square centre portions 35*a* of the guide members 35, in a direction opposite to the direction of rotation of the drive shaft 26. It will be understood, that by shifting the wedge-shaped members 28 in axial direction, the eccentricity of the first transmission member 34 will be adjusted, and consequently the ratio of transmission changed.

The guide pin members 35, 35*a* determine the circular track along which the first transmission member 34 rolls. A slippage is completely prevented due to the provision of the one-way coupling means 37, 38, 39.

The transmission member 34 is connected by a projecting ridge 41 to a disc 42 which has a corresponding slot in which the projecting ridge 41 is received. The disc 42 is provided at the rear face thereof with a slot 45*a* extending perpendicular to the ridge 41, and another disc 44 is provided with a ridge 45 engaging the slot 45*a*. The disc 44 is fixedly connected to the driven shaft 46 which is mounted by means of ball bearings 48 in the housing portion 23. Consequently it is possible to adjust the eccentricity of the transmission member 34 while maintaining a connection between the transmission member 34 and the driven shaft 46. The discs 42 and 44 operate in the manner of a universal joint.

As previously described, the adjusting member 28 is provided with wedge-shaped faces 28*b* and 28*c* in addition to the wedge-shaped face 28*a* which co-operates with the eccentric means 33. Counterbalancing means including two counterbalancing weights 49 and 50 are respectively mounted on the wedge-shaped faces 28*b* and 28*c*, and move in opposite radial directions when the adjusting member 28 is shifted in axial direction. Whenever the transmission member 34 is shifted to change its eccentricity, the counterbalancing means 49 and 50 are correspondingly shifted so that the weight of the transmission member 34 is counterbalanced in any of the adjusted positions of the transmission member 34. Consequently, all masses rotating with the shaft 26 are in static and dynamic equilibrium. The adjusting member includes in addition to the wedge-shaped members 28 also the member 30 which is turnably mounted by ball bearings 31 in a member 51 which is threaded on a tubular extension 52 of the housing portion 21. The member 51 is provided with a gear crown 53 which meshes with a first pinion 54 driven by another pinion 54 connected to a manually turned shaft 55. Members 55, 54, 53, 51 constitute operating means for moving the adjusting member 51 and 28 between its adjusted position which vary the eccentricity of the eccentric means 33.

The embodiment illustrated in Figures 5 to 6 obtains a complete counterbalancing of the masses which results in a quiet run of the transmission. However, this embodiment has the disadvantage that the transmission member is eccentric with respect to the driven shaft so that it is necessary to provide universal joint

means between the eccentric transmission member and the driven shaft to permit a variation of the eccentricity.

The embodiment of the present invention whose principle is illustrated in Figure 7 overcomes this disadvantage. In this embodiment of the present invention, the universal joint is eliminated, and another transmission arrangement according to the present invention is provided between the first transmission member and the driven shaft. Thereby, the transmission ratio is further reduced.

Referring now to Figure 7, two transmission members 56 and 57 are connected to each other for rotation, and may be made integral.

The transmission members 56 and 57 are connected by suitable eccentric means shown to be a crank portion of the drive shaft 58. Each of the transmission members 56 and 57 is provided along the circular periphery thereof with radial slots as described with reference to Figures 3 and 5. A stationary transmission member 59 provided with guide pin members, not shown in Fig. 7, co-operate with the transmission member 56. Consequently, transmission member 56 rolls along transmission member 59, and transmission member 57 turns with transmission member 56. The transmission member 57 co-operates with guide pin members, not shown in Fig. 7, which are mounted on a turnable transmission member 61 connected to the driven shaft 64. Since the transmission member 61 is turnable it is rotated by the turning transmission member 57 whereby driven shaft 64 is rotated. The transmission ratio between the transmission members 59 and 56 is $\frac{D-d}{D}$. The same ratio is valid for the transmission 57, 61 so that the transmission ratio of the entire arrangement can be expressed by the equation $\frac{D-d}{D.2}$. In these equations it is assumed that the diameters of the circles along which the guide pin members of members 59 and 61 are arranged, are of the same diameter, and it is also assumed that the slots of the transmission members 56 and 57 are arranged along circles having the same diameter. As previously explained, the difference between the diameter of the circles of the guide pin members D and the diameter of the inner transmission members 56 and 57 d is equal to $2e$, wherein e is the radial distance between the axis of shaft 58 and the axis of the crank portion about which the members 56 and 57 are turnable.

Assuming that the transmission ratio between the members 59 and 56 is 10:1 the transmission ratio of the entire transmission is 5:1. The eccentric masses 56 and 57 are counterbalanced by counterbalancing means 62 and 63 arranged eccentric with respect to the axis of the shaft 58. The transmission members 56 and 57 turn about the respective centres thereof in a direction opposite to the direction of rotation of drive shaft 58. Since such turning movement of the transmission member 57 is combined with the turning movement of the centre of transmission member 57 about the axis of the drive shaft 58, the transmission member 61 turns in a direction of rotation opposite to the direction of rotation of the drive shaft 58.

The embodiment illustrated in Figs. 8—11 operates on the principle described with reference to Fig. 7. The housing of the transmission includes a portion 65 having roller bearings 66 for the driven shaft 67 and for the disc 68 which is connected to shaft 67 for rotation therewith. The centre portion 69 of the housing encloses the rotary parts of the transmission. The other end portion of the housing 70 supports the drive shaft 71 in ball bearings 72. The free end of the drive shaft 71 is mounted by a roller bearing 72*a* in the disc 68.

The drive shaft 71 is provided with eccentric means 73, the eccentricity of which can be adjusted. The eccentric means 73 is shown to be a cylindrical member on which the transmission members 75*a* and 75*b* are mounted for turning movement. The transmission members 75*a* and 75*b* are connected by a cylindrical member and by screws 74. Roller means are arranged between the cylindrical portion of the transmission members 75*a* and 75*b* and the cylindrical member 73 of the eccentric means. The transmission members 75*a* and 75*b* correspond to the transmission members 56 and 57 described with reference to Fig. 7. Each of the transmission members 75*a* and 75*b* is provided along the circular periphery thereof with slots 75*c* engaging a set of square pins 76 which are slidable in the slots 75*c*. The cylindrical end portions of the guide pins 76 are mounted in pairs of one-way coupling members 87 which are arranged in circular guide grooves of members 77 and 78, and of members 79, 80. Balls 77*a* are provided in wedge spaces defined by the portions 77*b* of the one way slip coupling members 87 and co-operate with the walls of the associated circular grooves for blocking movement in one direction of rotation, while permitting movement in the opposite direction of rotation. Movement of the guide members 88 is blocked in a direction of rotation which is opposite the direction of the torque which is to be transmitted.

An adjusting member 82 is mounted on shaft 67 for movement in axial direction and is prevented from turning relatively thereto by a key 81 engaging corresponding key grooves in shaft 67 and in adjusting member 82. The outer surface of the adjusting member 82 is prismatic so that member 82 has a square cross section, as best seen in Fig. 9a.

The outer surface of member 82 is provided with grooves 83 which are composed of three relatively inclined portions. The portions of the grooves 83 are inclined to the axis of the drive shaft 67. As best seen in Figs. 11, 11a and 11b, the transmission member *75a*, *75b* is mounted on the centre portion of the adjusting member 82, the adjusting member 82 being located in a rectangular opening of eccentric cylinder 73 which turnably supports the transmission members *75a*, *75b*. An inclined key groove is provided in the rectangular opening of the transmission members *75a*, *75b* opposite the central portion of the groove 83. A spring key *83b* engages such oppositely arranged grooves. The counterbalancing means 84 and 85 are provided with central openings of rectangular cross section through which the end portions of the adjusting member 82 pass. Key grooves *83c* are provided in the rectangular openings of the counterbalancing means 84 and 85, and keys *83a* engage such grooves and the end portions of the groove 83 in the adjusting member 82. When the adjusting member 82 is shifted in axial direction, the radial position of the transmission members *75a*, *75b* and of the counterbalancing means 84, 85 is adjusted. One position is shown in Fig. 11, and an adjusted position is shown in Fig. 11a. It is apparent that shifting of the adjusting member 82 will displace the counterbalancing means 84, 85 in radial direction opposite to the shifting of the transmission members *75a* and *75b*.

When the drive shaft 71 is rotated, the transmission members *75a* and *75b* are turned about the axis of shaft 71, with the centres moving along a circle about the axis of shaft 71. At the same time, the transmission members *75a* and *75b* are forced by the guide members 88, 76 to turn about the eccentric means 73 in an opposite direction of rotation. Since the transmission member 79, 80 is turnable in the housing 69 due to the provision of roller 110, the rotary motion of the transmission member *75b* is transferred to the transmission member 79, 80. Screws 89 connect the transmission member 79, 80 with the disc 68 so that the driven shaft 87 is rotated as explained with reference to the schematic illustration of Fig. 7.

In order to shift the adjusting member 82 in axial direction, which is necessary for varying the transmission ratio of the device, an annular member 90 is mounted on member 82 and co-operates with an annular plate 91. Ball bearing means 92 are provided between the plate 91 and the member 93 which is provided with an outer thread 94 engaging corresponding threads 95 provided on the end portion of the housing 70. The annular body 93 has gear means 96 on the outer surface thereof which co-operates with a worm 97 so that by turning of worm 97 member 93 is turned for shifting the plate 91 in axial direction. Spring means 98 are arranged between the counterbalancing member 84 and the ring 90 which is connected to the adjusting member 82 for movement therewith. Consequently the spring means 98 urge the adjusting member 82 into the position illustrated on the left side of Fig. 8, corresponding to the smallest possible ratio of the transmission. When the adjusting member 82 is shifted in opposite axial direction by operation of the operating means 97, 96, 93 and 91, the spring 98 is compressed as shown on the right side of Fig. 8. This position corresponds to the maximum transmission ratio.

Fig. 11 and 11a clearly shew in a somewhat schematic illustration how the counterbalancing means 84, 85 are shifted in the direction opposite to the shifting of the member *75a*, *75b* when the adjusting member 82 is shifted in the axial direction. Due to the effect of the adjusting member of the eccentric means, the rotary elements are counterbalanced in each adjusted position.

From the above described description of several embodiments of the present invention, it will be apparent that the transmission according to the present invention has positively connected transmission members so that a slippage between the members is impossible but it is, nevertheless, gradually and infinitely adjustable. Assuming 1500 revolutions per minute of the drive shaft, it is possible to vary the number of revolutions per minute of the driven shaft between standstill and 200 revolutions per minute. Such variation of the transmission ratio is obtained by operating the operating means and by varying the eccentricity of the respective transmission member as described above.

It will be understood that each of the elements described above, or two or more together, may also find a useful application in other types of transmissions differing from the types described above.

While the invention has been illustrated and described as embodied in a variable transmission arrangement including a rotating eccentric transmission member and another rotary transmission member rolling on the first-mentioned transmission member, it is not intended to be limited to the details shewn, since various modifications and structural changes may be made without departing in any way from the scope of the present invention.

WHAT I CLAIM IS:—

1. A variable speed torque transmission arrangement of the kind hereinbefore set forth characterized by the provision of rotatable transmission means having a plurality of guide slots evenly spaced around a pitch circle and extending radially in relation to said pitch circle, and guide means located in each guide slot, the said guide slots and guide means co-operating to impart rotational movement to the rotatable transmission means about an axis

coincident with the centre of the pitch circle in a direction opposite to the direction of the traversing movement of the rotatable transmission means around the circular path.

2. An arrangement according to Claim 1, characterised in that the guide means are mounted in spaced relationship in an annular groove or the like, said guide means having associated locking means arranged so that the guide means can move only in one direction along the groove but which in the other direction prevent the rotatable transmission means moving relatively to the circular path.

3. An arrangement according to Claim 1 or 2, characterised by the provision of a driving and a driven shaft associated with the rotatable transmission means and that on the driving or driven shaft preferably two counterbalance masses are staggered eccentrically and in parallel planes by 180° in relation to one another or in relation to the rotatable transmission means in such manner that both the eccentricity of said counterbalance masses and that of the rotatable transmission means are adjustable simultaneously and in dependency on one another.

4. An arrangement according to Claim 3 characterised in that the simultaneous adjustment is effected by means of wedges, which are displaceable on the driving shaft and on which are disposed three eccentric surfaces which are disposed one above the other at a corresponding height and which are limited in the form of a wedge and on which the rotatable transmission means and the two counterbalance masses rotate with correspondingly wedge-shaped surfaces.

5. An arrangement according to Claim 4, characterised in that the axially displaceable wedges each have, on the end remote from the rotatable transmission means, a nose, the noses being embraced by a sleeve guided in the bottom part of a housing and displaceable in the axial direction by means of a toothed drive.

6. An arrangement according to Claim 1, characterised in that the rotatable transmission means consists of two discs disposed co-axially with one another, one of said discs having an associated stationary annular body and the other disc an associated annular body which is rotatable and connected to a driving shaft.

7. An arrangement as claimed in Claim 6, characterised in that the driving shaft is disposed in a housing and has a bush with three eccentrics which are disposed one above the other, which carries two counterbalance masses and a double driven disc in an eccentric mounting, the bush being provided, in the range of adjustment of the three eccentrics, with grooves which extend at an angle and in which slide spring wedges adjusting the eccentric stroke, so that by axial displacement of the bush in relation to the shaft different deflections or adjustments of the eccentrics are obtained.

8. An arrangement as claimed in Claim 7, characterised in that on the driving side the bush has a collar, over which engages a plate which is axially adjustable by means of a screw-thread and which raises the bush against the pressure of a spring into the position of maximum speed of revolution or lowers said bush against the pressure of said spring into the position of minimum speed of revolution.

9. A variable speed torque transmission arrangement according to any of the embodiments herein described with reference to Figs. 3 and 4, or Figs. 5, 5a, 5b, 5c and 6, or Figs. 7 to 11 of the accompanying drawings.

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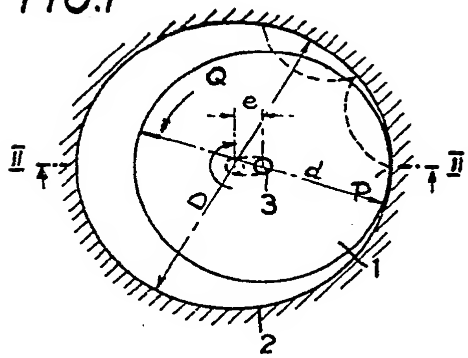

FIG.1

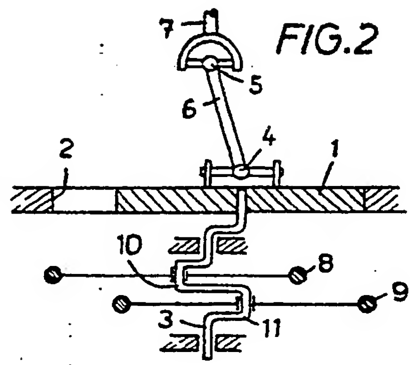

FIG.2

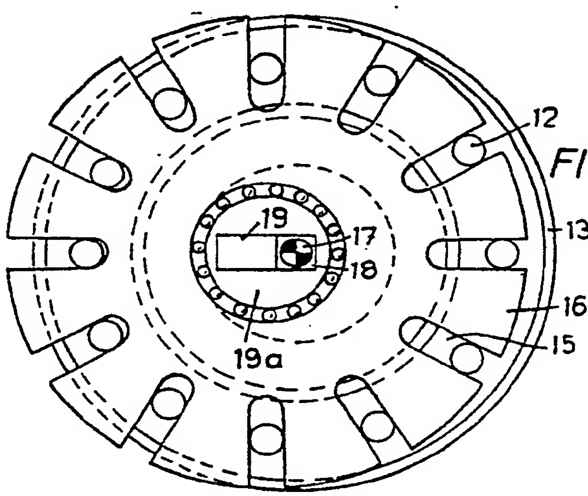

FIG.3

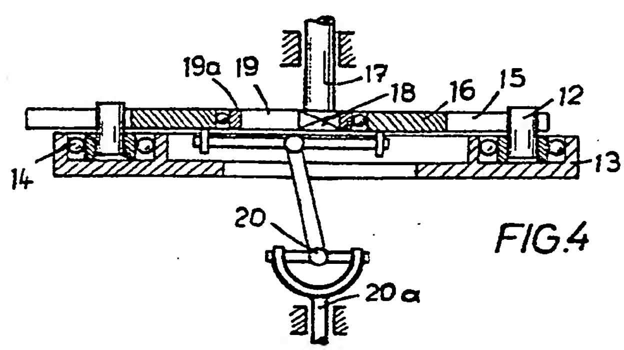

FIG.4

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8 SHEETS

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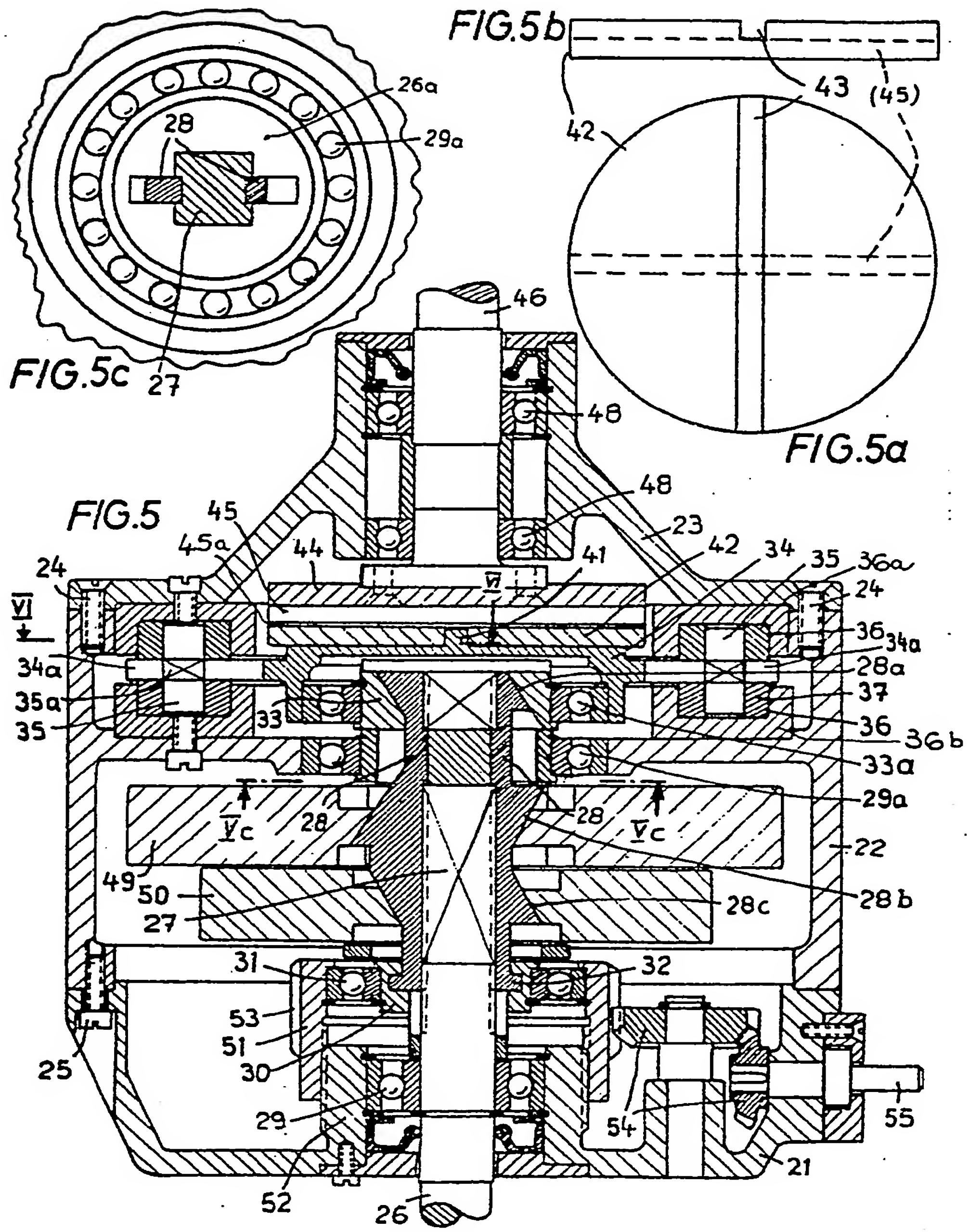
SHEETS 1 & 2



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SHEETS 1 & 2

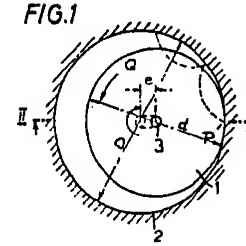


FIG.1

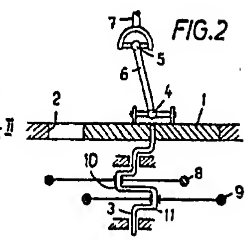


FIG.2

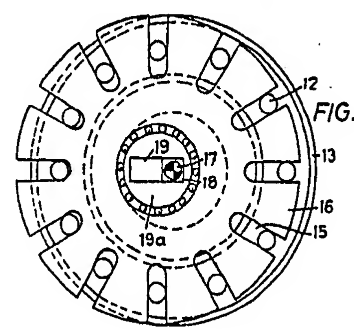


FIG.3

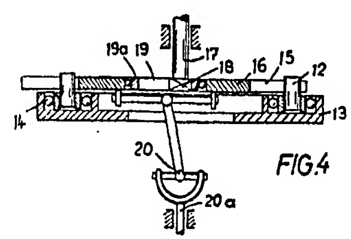


FIG.4

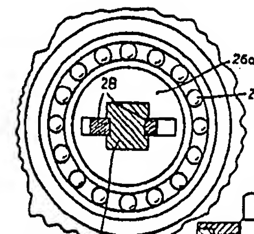


FIG.5c

FIG.5b

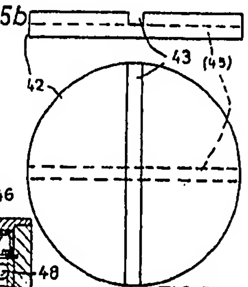


FIG.5a

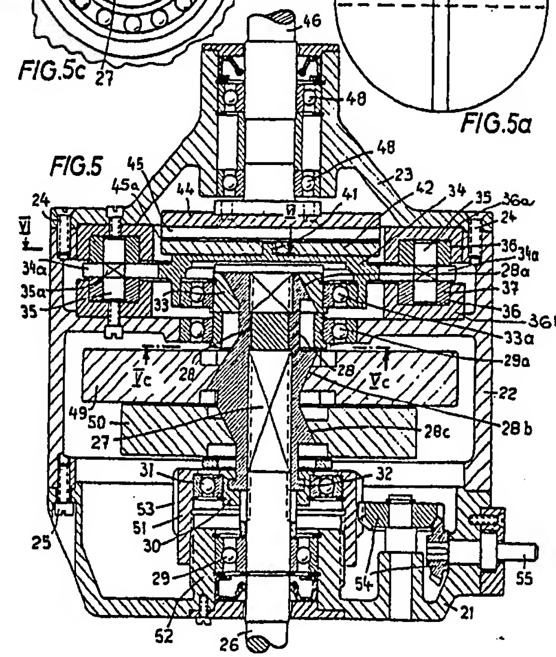


FIG.5

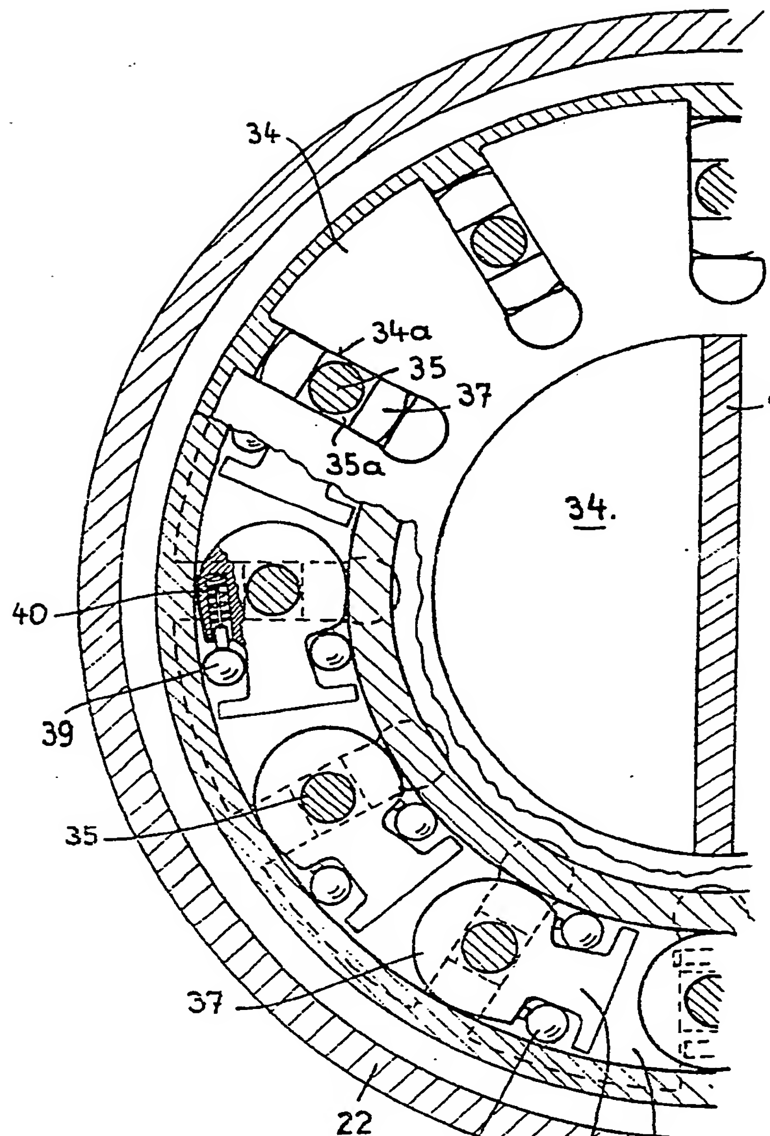
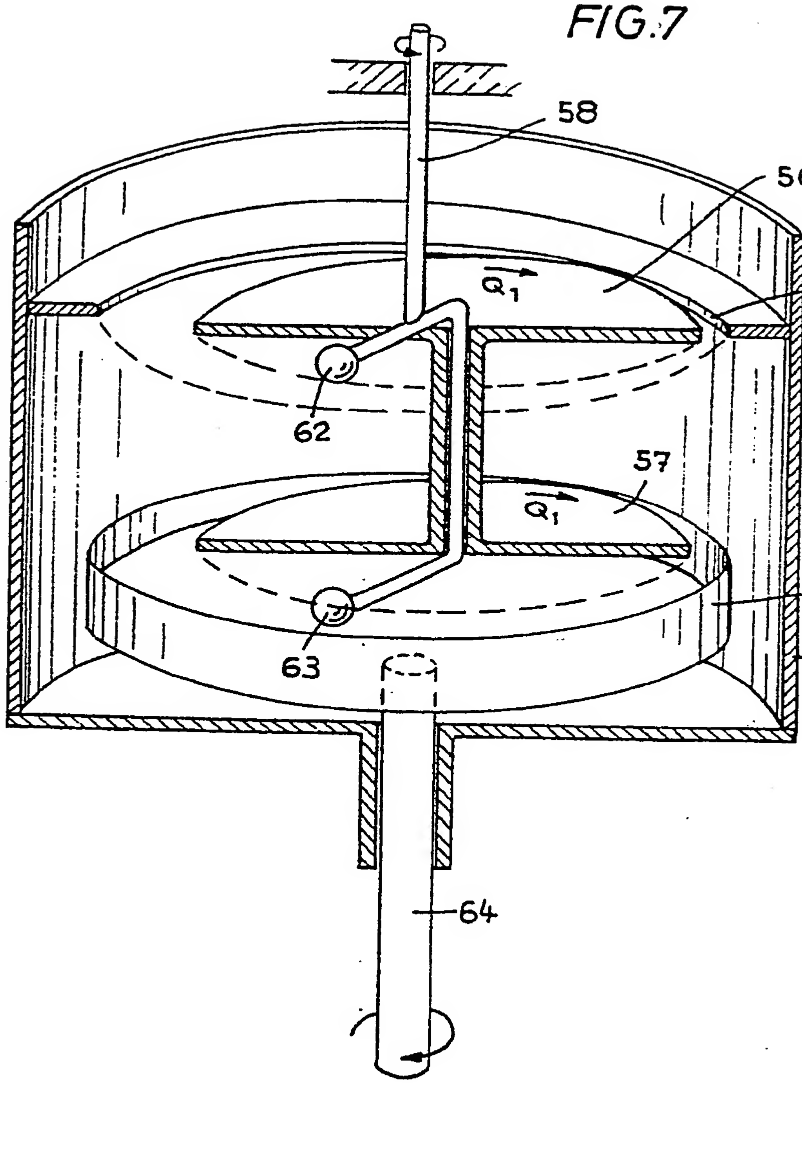


FIG.6

'G.6

FIG.7



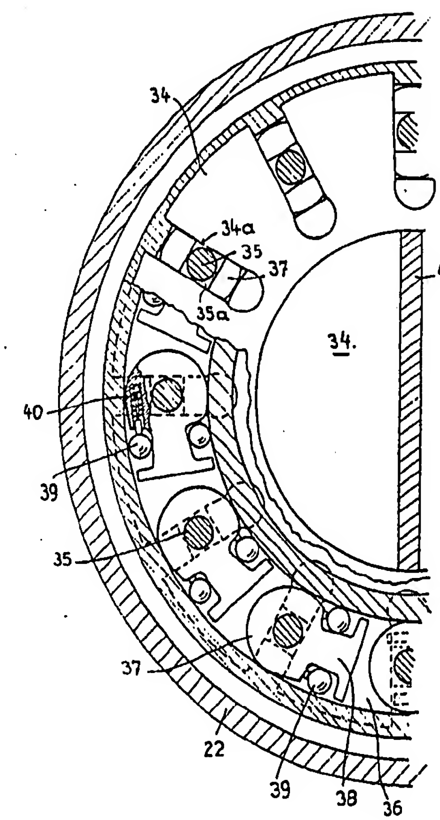

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FIG.6

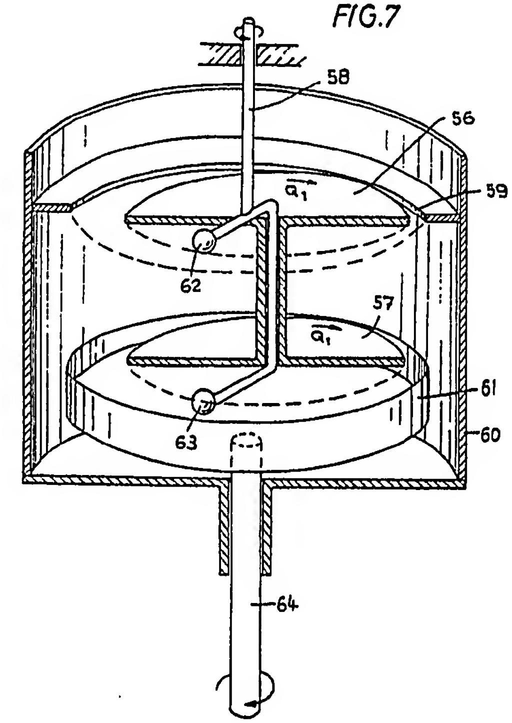
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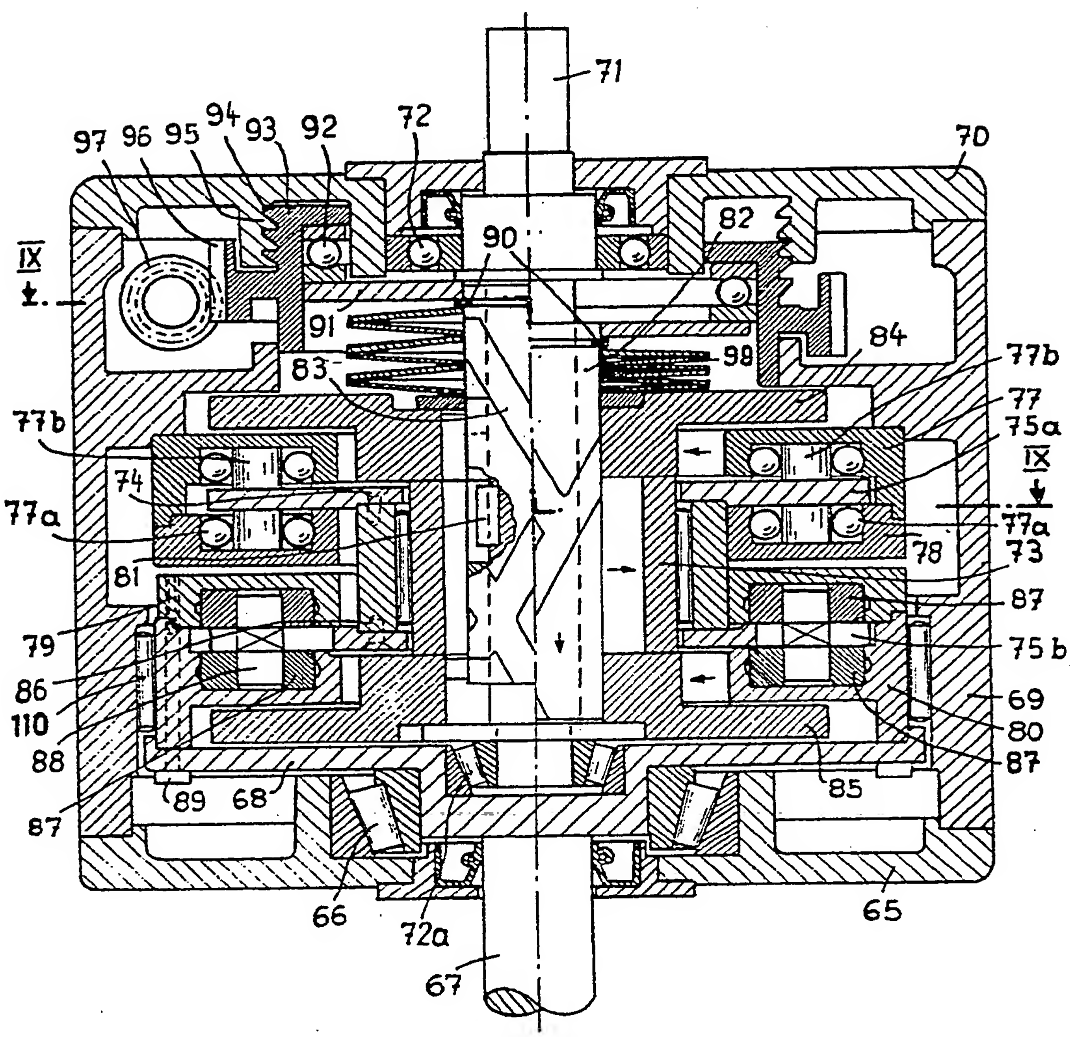
FIG.7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75a
IX
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78
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75b
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110
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72a
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FIG.8

821,857 COMPLETE SPECIFICATION
8 SHEETS
This drawing is a reproduction of the Original on a reduced scale.
SHEETS 5 & 6

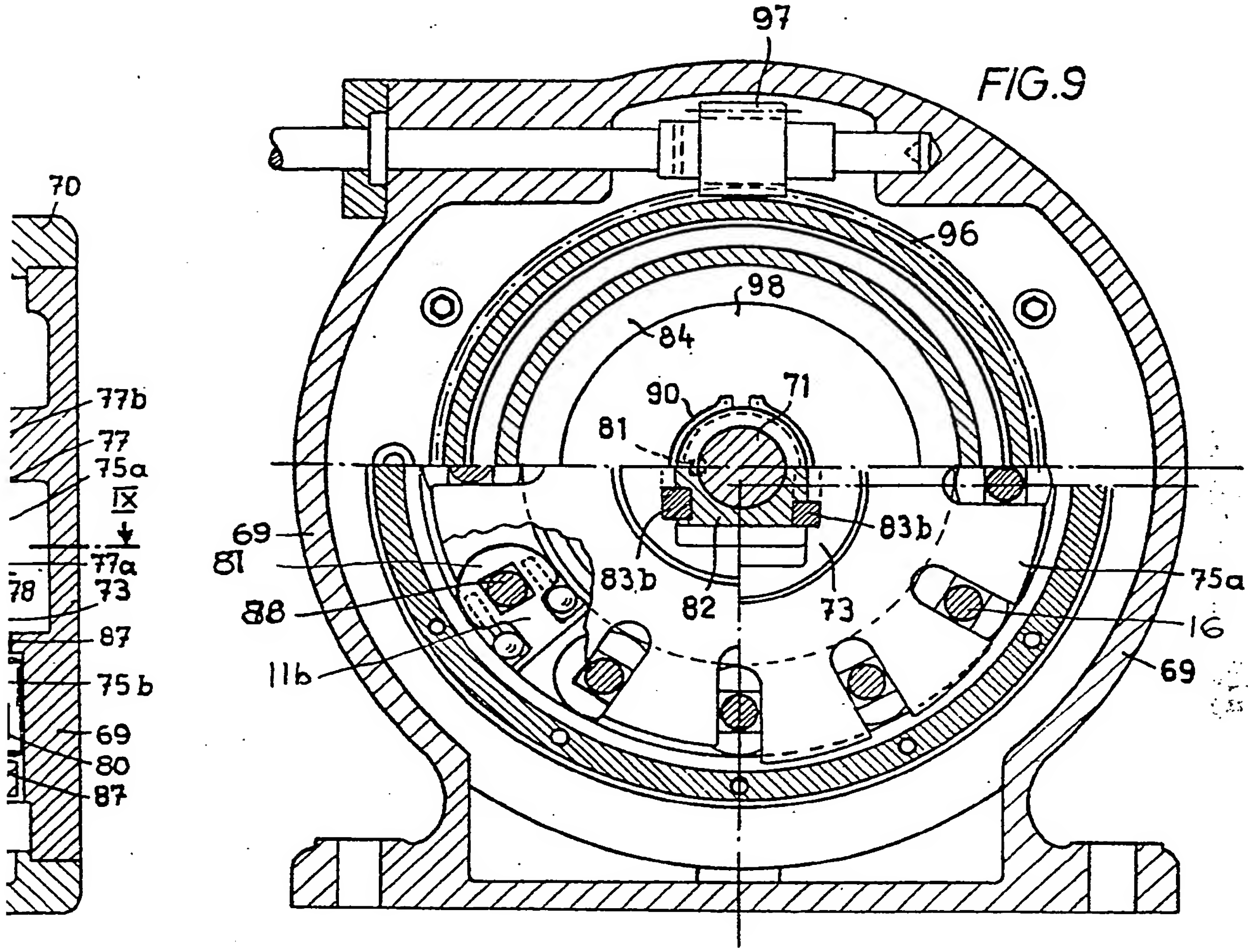


FIG.9

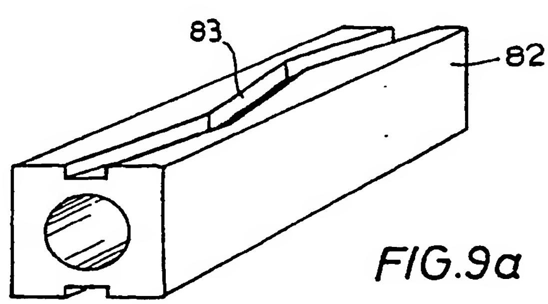


FIG.9a

821,857 COMPLETE SPECIFICATION
8 SHEETS This drawing is a reproduction of the Original on a reduced scale.
SHEETS 5 & 6

FIG.8

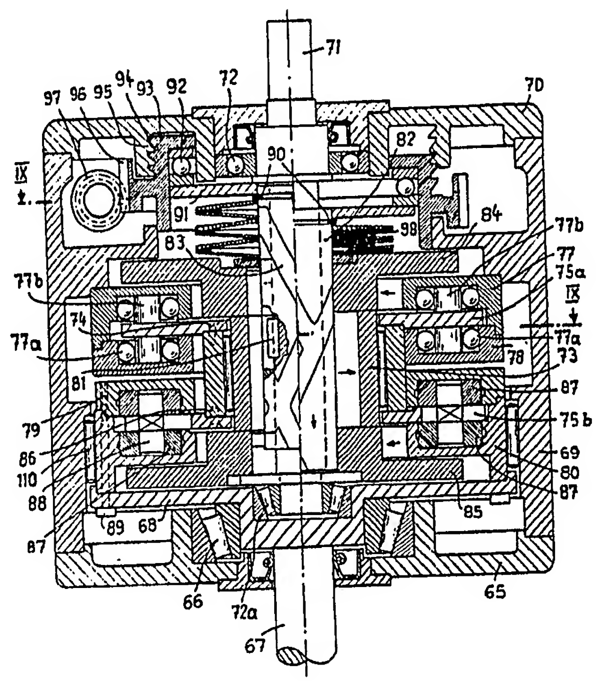
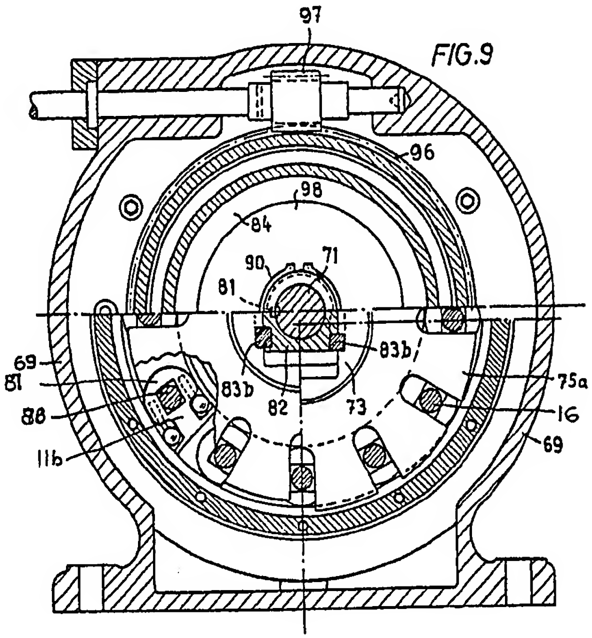


FIG.9



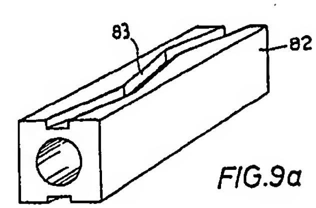
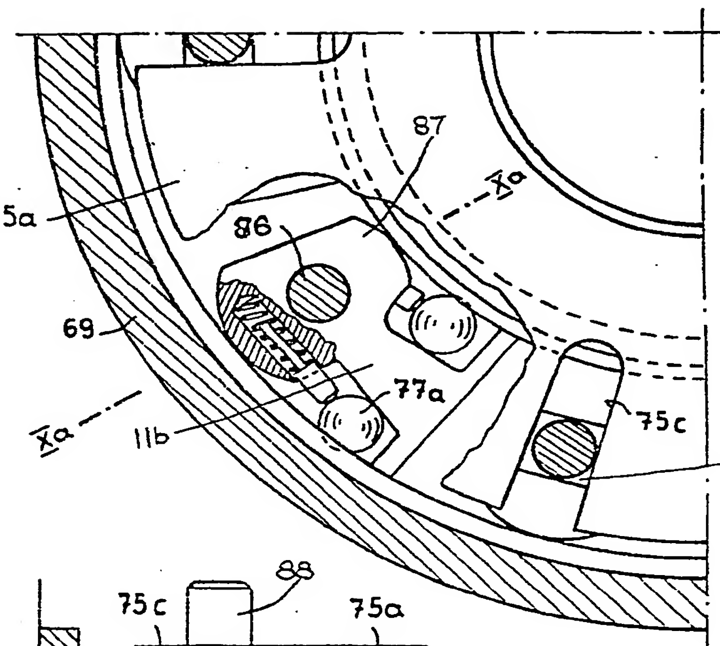


FIG.9a

FIG.10



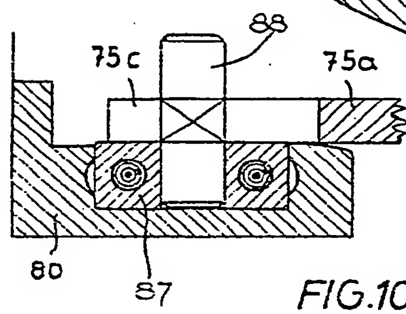


FIG.10a

821,857 COMPLETE SPECIFICATION
8 SHEETS

This drawing is a reproduction of the Original on a reduced scale.
SHEETS 7 & 8

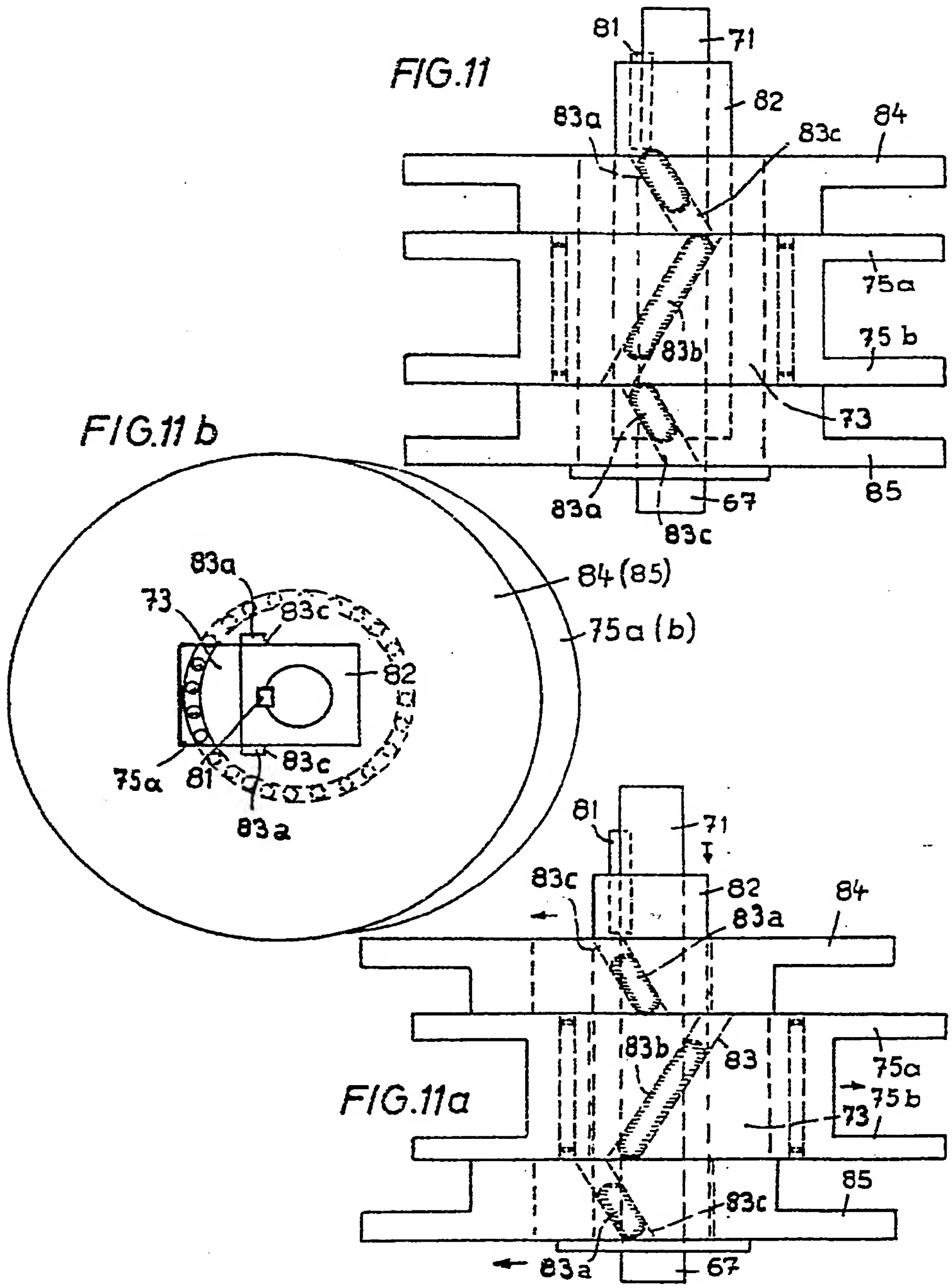


FIG.11

FIG.11 b

FIG.11a

76

821,857 COMPLETE SPECIFICATION
SHEETS This drawing is a reproduction of the Original on a reduced scale. SHEETS 7 & 8

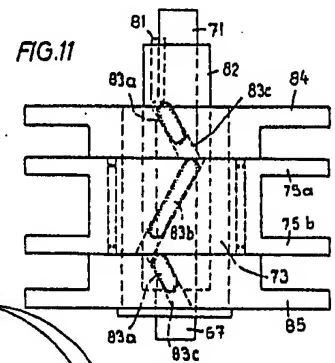


FIG.11

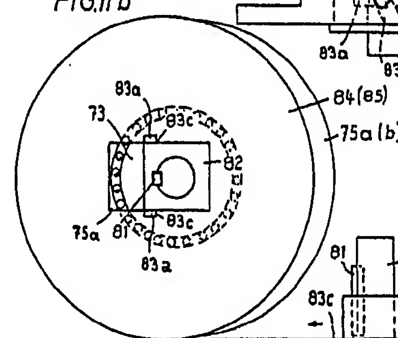


FIG.11 b

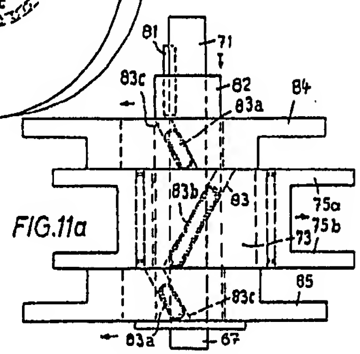
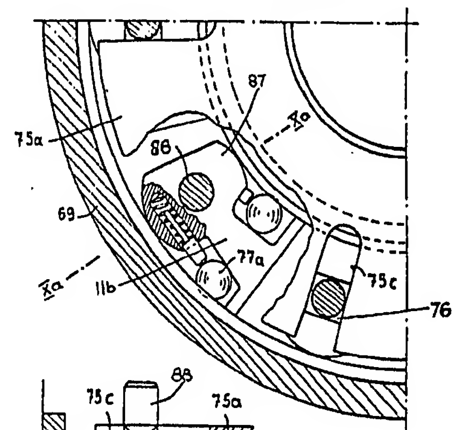


FIG.11a

FIG.10



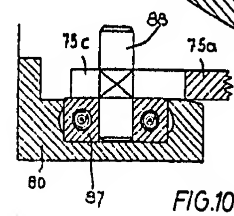


FIG.10a